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DESCRIPTION

WHEEL SUPPORTING ROLLING BEARING UNIT

<Technical Field>

The present invention relates to improvement in a wheel supporting rolling bearing unit for supporting rotatably a wheel, especially an idler wheel (a rear wheel of an FF car, a front wheel of an FR car and an RR car), on a suspension system of a vehicle (car).

<Background Art>

As the wheel supporting rolling bearing unit, structures shown in FIGS.11 and 12 are set forth in JP-A-2001-221243, for example. First, a structure of a first example shown in FIG.11 will be explained hereunder. A wheel 1 constituting the wheel is supported rotatably on an end portion of an axle 3 constituting a suspension system by a wheel supporting rolling bearing unit 2. More particularly, inner rings 5, 5 as a stationary side raceway ring, which constitutes the wheel supporting rolling bearing unit 2, are fitted onto a supporting shaft 4 fixed to the end portion of the axle 3, and then the inner rings 5, 5 are fixed with a nut 6. Meanwhile, the wheel 1 is coupled/fixed to a hub 7 as a rotary side raceway ring, which constitutes the wheel supporting rolling bearing unit 2, by a plurality of stud bolts 8, 8 and nuts 9, 9.

Double row outer ring raceways 10a, 10b that act as a

rotary side raceway surface respectively are formed on an inner peripheral surface of the hub 7, and a fitting flange 11 is formed on an outer peripheral surface of the same. The wheel 1 as well as a drum 12 constituting a baking system is coupled/fixed to a one-side surface (an outside surface in the first example, a left side surface in FIGS.11 and 12) of the fitting flange 11 by the stud bolts 8, 8 and nuts 9, 9.

Balls 14, 14 are provided rollably between the outer ring raceways 10a, 10b and inner ring raceways 13, 13, which are formed on outer peripheral surfaces of the inner rings 5, 5 to act as the stationary side raceway surface respectively, every plural pieces in a state that these balls 14, 14 are held in cages 15, 15 respectively. A double row angular contact ball bearing having a back-to-back arrangement is constructed by combining respective constituent members mutually in this manner, so that the hub 7 is supported rotatably around the inner rings 5, 5 to bear the radial load and the thrust load. In this case, seal rings 16a, 16b are provided between inner peripheral surfaces of both end portions of the hub 7 and outer peripheral surfaces of end portions of the inner rings 5, 5 to isolate a space in which the balls 14, 14 are provided from the external space. In addition, an opening portion of an outer end (Here, an "outside in the axial direction" means the outside of the hub in the width direction when such hub is fitted to the vehicle. Similarly, an "inside" means the

inside of the hub on the middle side in the width direction.) of the hub 7 is covered with a cap 17. The cap 17 has no portion that slidingly comes into contact with the inner rings 5, 5 as the other raceway ring and the axle 3 and the nut 6 that are stationary portions with respect to the inner rings 5, 5.

Upon using the above wheel supporting rolling bearing unit 2, as shown in FIG.11, the supporting shaft 4 onto which the inner rings 5, 5 are fitted and fixed is fixed to the axle 3 and also the wheel 1, to which a tire (not shown) is fitted, and the drum 12 are fixed to the fitting flange 11 of the hub 7. Also, a braking drum brake is constructed by assembling in combination the drum 12, a wheel cylinder and shoes (not shown) supported on a backing plate 18 fixed to the end portion of the axle 3. Upon braking, a pair of shoes provided to the inner diameter side of the drum 12 are pushed against an inner peripheral surface of the drum 12.

Next, a structure of a second example shown in FIG.12 in the prior art will be explained hereunder. In the case of this wheel supporting rolling bearing unit 2a, a hub 7a as the rotary side raceway ring is supported rotatably by a plurality of balls 14, 14 on the inner diameter side of an outer ring 19 as the stationary side raceway ring. Therefore, the double row outer ring raceways 10a, 10b as the stationary side raceway surface are provided on the inner peripheral

surface of the outer ring 19 respectively, and first and second inner ring raceways 20, 21 as the rotary side raceway surface are provided on the outer peripheral surface of the hub 7a respectively.

The hub 7a is constructed by using a hub main body 22 as a main shaft member and an inner ring 23 in combination. A fitting flange 11a for supporting the wheel is provided to an outer end portion of the outer peripheral surface of the hub main body 22, and the first inner ring raceway 20 is provided to an middle portion thereof, and also a small-diameter stepped portion 24 that is smaller in diameter than a portion in which the first inner ring raceway 20 is formed is provided to the middle portion thereof near an inner end. Then, the inner ring 23, to an outer peripheral surface of which the second inner ring raceway 21 having a circular-arc sectional shape is provided, is fitted onto the small-diameter stepped portion 24. In addition, an inner end surface of the inner ring 23 is pressed with a caulking portion 25 that is formed by elastically deforming an inner end surface of the hub main body 22 outward in the radial direction. Thus, the inner ring 23 is fixed to the hub main body 22. Further, seal rings 16c, 16d are provided between an inner peripheral surface of the outer ring 19 on both end portions and the outer peripheral surface of the middle portion of the hub 7a and the outer peripheral surface of the inner end portion of the inner ring

23 respectively. Thus, the spaces in which the balls 14, 14 are provided are isolated from the external space between the inner peripheral surface of the outer ring 19 and the outer peripheral surface of the hub 7a.

In this case, in the case of the above wheel supporting rolling bearing unit 2a shown in FIG.12, the rigidity can be enhanced because the first inner ring raceway 20 is formed directly on the outer peripheral surface of the middle portion of the hub main body 22. In other words, the first inner ring raceway to be provided in the middle portion of the wheel supporting rolling bearing unit can be formed on the outer peripheral surface of the inner ring prepared as a separate body of the hub main body, and then this inner ring can be fitted/fixed onto the hub main body. In this case, unless an inference of the inner ring into the hub main body is increased, the rigidity is lowered, like a structure shown in FIG.12, rather than the case where the first inner ring raceway 20 is formed directly on the outer peripheral surface of the middle portion of the hub main body 22. A working of fitting the inner ring as the separate body onto the hub main body from the inner end portion to the middle portion while keeping a large inference is troublesome. In contrast, as shown in FIG.12, the wheel supporting rolling bearing unit 2a having the high rigidity can be constructed without trouble by employing the structure in which the first inner ring raceway

20 is formed directly on the outer peripheral surface of the middle portion of the hub main body 22.

In the case of the above conventional structure, it is unavoidable that a torque (running resistance of the wheel supporting rolling bearing unit) required to turn the hub 7 (or 7a) is increased because the seal rings 16a, 16b (or 16c, 16d) are provided to the opening portions on both ends of the internal space in which the balls 14, 14 are provided. Meanwhile, for example, as set forth in JP-A-2001-241450, the structure in which, in order to isolate the internal spaces from the external space, the one end side of the internal space is closed by the cap and also the seal ring is provided only to the other end side in the axial direction is well known in the prior art.

However, in the case of the wheel supporting rolling bearing unit set forth in JP-A-2001-241450, because the running resistance of the seal ring is not always low, it is impossible to sufficiently reduce the rolling resistance of the wheel supporting rolling bearing unit. As a result, since the running performances, mainly the acceleration performance and the fuel consumption performance, of the vehicle into which the wheel supporting rolling bearing unit is incorporated become worsen, improvement in the running performances is desired in view of the recent trend toward the energy saving. The technology to reduce the sliding resistance between the

seal member and the sliding portion of the counter member by mixing plastic fine grains which are impregnated with the lubricant into the rubber composition constituting the seal member is known in JP-A-8- 319379. However, no description is given in JP-A-8-319379 that suggests getting of the high-performance structure as a whole by applying the above rubber composition to the wheel supporting rolling bearing unit.

In addition, as the structure that reduced the running torque of the rolling bearing unit by reducing the resistance of the seal-ring providing portion, improvement in the inference of the seal lip as set forth in JP-A-10-252762 was considered in the prior art.

In the case of the wheel supporting rolling bearing unit as the subject of the present invention, even though the running torque should be reduced, the structure capable of keeping the wheel supporting rigidity to ensure the controllability and also preventing sufficiently the entering of the foreign matter into the internal space of the rolling bearing unit to ensure the durability of the rolling bearing unit is needed. In other words, the supporting rigidity must be assured by enhancing the rigidity of the rolling bearing unit to ensure the controllability, nevertheless the rolling resistance of respective rolling members is increased if a preload applied to respective rolling members is increased simply to enhance

the rigidity, whereby the running torque cannot be reduced. Also, in case it is considered only that the sliding resistance of the seal ring is lowered simply, prevention of the entering of the foreign matter into the internal space of the rolling bearing unit cannot be sufficiently attained and thus the durability cannot be sufficiently assured.

A wheel supporting rolling bearing unit of the present invention has been made in view of such circumstances, and realizes a structure that has a high rigidity, an excellent durability, and a low running torque.

<Disclosure of the Invention>

A wheel supporting rolling bearing unit of the present invention comprises a stationary side raceway ring, a rotary side raceway ring, a plurality of balls, and a seal ring, as in the above wheel supporting rolling bearing unit known in the prior art.

The stationary side raceway ring is supported/fixed on a suspension system in use.

The rotary side raceway ring supports/fixes a wheel in use.

A plurality of balls are provided between a stationary side raceway surface and a rotary side raceway surface, each of which has a circular-arc sectional shape, on mutually opposing peripheral surfaces of the stationary side raceway ring and the rotary side raceway ring.

The seal ring seals only one opening portion out of opening portions on both end portions of a space, in which the balls are provided, between the mutually opposing peripheral surfaces of the stationary side raceway ring and the rotary side raceway ring.

The other raceway ring, which is positioned inside in a radial direction, out of the stationary side raceway ring and the rotary side raceway ring consists of a main shaft member and an inner ring. Then, the main shaft member has a first inner ring raceway formed directly in a middle portion of an outer peripheral surface in an axial direction to serve as the stationary side raceway surface or the rotary side raceway surface and a small-diameter stepped portion formed on one end portion of the outer peripheral surface in the axial direction. Also, the inner ring on an outer peripheral surface of which a second inner ring raceway as the stationary side raceway surface or the rotary side raceway surface is formed is fitted/fixed onto the small-diameter stepped portion.

The seal ring has two or three seal lips which are formed of elastic material respectively and a top end edge of each of which slidingly comes into contact with a counter surface.

In particular, in the wheel supporting rolling bearing unit of the present invention, an axial load to apply a preload to the balls is set to 1.96 to 4.9 kN.

Also, a rigidity factor is set to 0.09 or more.

Also, a torque required to relatively run the stationary side raceway ring and the rotary side raceway ring at 200 min^{-1} (200 RPM) based on a friction between the seal lip and a counter surface is set to 0.03 to 0.2 N·m.

In addition, a torque required to relatively run the stationary side raceway ring and the rotary side raceway ring at 200 min^{-1} based on a rolling resistance of the balls is set to 0.15 to 0.45 N·m.

The rigidity factor set forth in this specification means a ratio (R/Cr) of the rigidity R [kN·m/deg] of the wheel supporting rolling bearing unit to a radial dynamic rated load Cr [N] of the wheel supporting rolling bearing unit. The rigidity R in this case is represented by an inclination angle between both raceway rings when a moment load is loaded to the rotary side raceway ring in a situation that the stationary side raceway ring constituting the wheel supporting rolling bearing unit is fixed. For example, the rigidity is measured as shown in FIG.13. In this case, FIG.13 shows the condition in which the rigidity R of the wheel supporting rolling bearing unit 2a shown in above FIG.12 is measured.

In the measuring operation, the outer ring 19 as the stationary side raceway ring is secured to an upper surface of a fixing pedestal 26 and a base end portion (left end portion in FIG.13) of a lever plate 27 is coupled/fixed to the fitting flange 11a of the hub 7a as the rotary side raceway ring. Then,

the load is applied to the portion, which is part from a center of rotation of the hub 7a by a distance L corresponding to a radius of rotation of the tire, for example, on the upper surface of the lever plate 27 to apply the moment load of 1.5 kN·m to the hub 7a via the lever plate 27. Since the hub 7a is inclined from the outer ring 19 based on this moment load, this inclination angle is measured as an inclination angle [deg] of a clamp face 29 of the fitting flange 11a to an upper surface 28 of the fixing pedestal 26. Then, the rigidity R [kN·m/deg] is derived by dividing the moment load (1.5 kN·m) by this inclination angle. Then, the above rigidity factor is derived by dividing the rigidity R by the radial dynamic rated load Cr [N] of the wheel supporting rolling bearing unit 2a.

In the case of the wheel supporting rolling bearing unit of the present invention constructed as above, the running torque can be reduced sufficiently while assuring the necessary rigidity and the durability.

First, since the axial load to apply a preload to the balls is regulated in a range of 1.96 to 4.9 kN, the running torque can be reduced sufficiently while maintaining the rigidity and the durability. If the axial load is below 1.96 kN, the preload is not enough and then the rigidity of the wheel supporting rolling bearing unit becomes insufficient. Thus, the controllability of the vehicle into which the wheel

supporting rolling bearing unit is incorporated becomes worse.

In contrast, if the axial load is in excess of 4.9 kN, the preload is applied excessively (a contact pressure of the rolling contact portion is increased excessively) and then the rolling resistance (running torque) of the wheel supporting rolling bearing unit is increased excessively. Then, a heating value at the rolling contact portion becomes too large, and then a temperature rise in the wheel supporting rolling bearing unit becomes conspicuous. Thus, the grease that is sealed in the bearing unit is ready to deteriorate immediately. As a result, the durability of the wheel supporting rolling bearing unit is lowered. Also, as the result of excessive increase of the contact pressure of the rolling contact portion, the rolling fatigue life of the stationary side raceway ring, the rotary side raceway ring and rolling contact surfaces of the balls is shortened, and the durability of the wheel supporting rolling bearing unit is lowered in this respect.

In contrast, in the case of the present invention, since the axial load to apply the preload to the balls is regulated in a range of 1.96 to 4.9 kN, the running torque can be reduced sufficiently while keeping the rigidity and the durability.

Also, since the rigidity factor is set to 0.09 or more, the controllability of the vehicle into which the wheel supporting rolling bearing unit is incorporated can be secured while maintaining the rigidity of the wheel supporting rolling

bearing unit. Conversely speaking, the controllability becomes worse if the rigidity factor is below 0.09.

Here, because the higher rigidity factor is preferable from a viewpoint of assuring the controllability, an upper limit is not particularly defined. Any high value of the rigidity factor may be employed if other requirements are satisfied. Meanwhile, as the approach that is normally taken to enhance the rigidity factor, a value of the preload should be increased or a pitch diameter of the balls or a pitch between the balls arranged in double rows in the axial direction should be increased.

In this case, as described above, there is a limit to an increase of the preload. Also, there is a limit to increases of the pitch diameter and the pitch in the axial direction from a viewpoint of reduction in size and weight. Therefore, when the wheel supporting rolling bearing unit is made of normal steel material (the outer ring and the hub are made of S53C and the inner ring and the balls are made of SUJ2), an upper limit of the rigidity factor becomes almost 0.18. In this case, when a part (e.g., balls) or all of constituent elements of the wheel supporting rolling bearing unit are made of ceramics, the rigidity factor can be increased unless the preload and the pitch diameter and the pitch in the axial direction are increased. Therefore, in this case, the rigidity factor can also be increased to exceed 0.18.

Also, since the torque required to relatively run the stationary side raceway ring and the rotary side raceway ring at 200 min^{-1} based on the friction between the seal lip and the counter surface is set to 0.03 to 0.2 N·m, the running torque can be reduced sufficiently while assuring the durability of the wheel supporting rolling bearing unit.

That is, as the result of experiments made by the inventors of the present invention, it was understood that, insofar as the number of the seal lips is set to two or three, a validity of the sealing performance can be decided based on a magnitude of the running resistance of the seal ring irrespective of the structure of the seal ring. At the same time, it was understood that, if the running resistance of the seal ring is set to 0.03 N·m or more, the required sealing performance can be attained.

In the case of the wheel supporting rolling bearing unit of the present invention, since the torque of 0.03 N·m or more is assured, the sealing performance attained by the seal ring can be sufficiently assured by maintaining sufficiently the contact pressure of the slide contact portion between top ends of the seal lips constituting the seal ring and the counter surface. As a result, the durability of this wheel supporting rolling bearing unit can be ensured by preventing effectively the entering of the foreign matter such as a muddy water, or the like into the inside of the wheel supporting rolling bearing

unit. Conversely speaking, if the contact pressure of the slide contact portion between the top ends of the seal lips and the counter surface to reduce the torque lower than 0.03 N·m, a function of preventing the entering of the foreign matter become insufficient and thus the durability of the wheel supporting rolling bearing unit is lowered.

In contrast, if the torque exceeds 0.2 N·m, it is difficult to suppress the running torque of the wheel supporting rolling bearing unit as a whole sufficiently low (0.65 N·m or less).

On the contrary, in the case of the present invention, since the torque required to relatively run the stationary side raceway ring and the rotary side raceway ring at 200 min^{-1} based on the friction between the seal lip and the counter surface is regulated in a range of 0.03 to 0.2 N·m, the running torque can be reduced while assuring the durability.

In addition, since the torque required to relatively run the stationary side raceway ring and the rotary side raceway ring at 200 min^{-1} based on the rolling resistance of the balls is regulated in a range of 0.15 to 0.45 N·m, the running torque of the wheel supporting rolling bearing unit as a whole can be suppressed sufficiently low (0.65 N·m or less) while assuring the controllability and the durability.

If the torque is lessened below 0.15 N·m, the preload must be lowered considerably and, as described above, the

rigidity of the wheel supporting rolling bearing unit is not enough. Thus, the controllability of the vehicle into which the wheel supporting rolling bearing unit is incorporated is worsened.

In contrast, if the torque is enhanced so as to exceed 0.45 N·m, the increase of the preload is brought about. Thus, as described above, reduction in the durability of the wheel supporting rolling bearing unit is caused owing to the deterioration of the grease and the reduction in the rolling fatigue life, which are caused with an increase of the heating value at the rolling contact portion. Also, it is difficult to suppress the running torque of the wheel supporting rolling bearing unit as a whole sufficiently low.

In contrast, in the case of the present invention, since the torque required to relatively run the stationary side raceway ring and the rotary side raceway ring at 200 min^{-1} based on the rolling resistance of the balls is regulated in a range of 0.15 to 0.45 N·m, the running torque can be reduced while assuring the controllability and the durability.

<Brief Description of the Drawings>

FIG.1 is a sectional view showing a first example of a structure serving as a subject of the present invention;

FIG.2 is a sectional view showing a second example of the same;

FIG.3 is a half-plane sectional view showing a third

example of the same;

FIG.4 is a half-plane sectional view showing a fourth example of the same;

FIG.5 is a partial sectional view showing a first example of a particular structure of a seal ring, which can be applied to the present invention;

FIG.6 is a partial sectional view showing a second example of the same;

FIG.7 is a partial sectional view showing a third example of the same;

FIG.8 is a partial sectional view showing a fourth example of the same;

FIG.9 is a partial sectional view showing a fifth example of the same;

FIG.10 is a partial sectional view showing a first example of a structure that can reduce a slide resistance;

FIG.11 is a sectional view showing a first example of a wheel supporting rolling bearing unit, which is well known in the prior art, in a state that such unit is fitted into the suspension system;

FIG.12 is a sectional view showing a second example of the same; and

FIG.13 is a sectional view showing a condition in which a rigidity of the wheel supporting rolling bearing unit is measured.

<Best Mode for Carrying Out the Invention>

First, four examples of a structure of a wheel supporting rolling bearing unit as the subject of the present invention will be explained hereinafter. At first, FIG.1 shows a structure, as a first example, in which the running torque can be reduced readily by introducing improvements into the above structure shown in FIG.11 while assuring the sealing performance and the controllability. For this purpose, in the case of the present example, an inside inner ring raceway 13a is formed directly on an outer peripheral surface of the middle portion of a supporting shaft 4a as a main shaft member. As a result, the structure in which not only the rigidity can be assured not to render the manufacturing operation troublesome but also the foreign matters can be prevented from entering through the fitting portion between the inside inner ring 5 and the supporting shaft 4, already considered in the conventional structure shown in FIG.11, can be obtained. Also, the outside seal ring 16a incorporated into the conventional structure shown in FIG.11 is removed, and only the inside seal ring 16b is provided.

In the case where the present invention is applied to such structure, an axial load to apply the preload to the balls 14, 14 is set to 1.96 to 4.9 kN by regulating a torque that is applied to tighten the nut 6 being screwed onto the outer end portion of the supporting shaft 4a. Then, a torque (rolling

resistance) that is required to turn the hub 7 around the supporting shaft 4a at 200 min^{-1} is set to 0.15 to 0.45 N·m. This torque is increased as the axial load is increased if other specifications are set similarly. Therefore, in order to set the torque close to a lower limit (0.15 N·m), the axial load is set close to a lower limit (1.96 kN). When the torque is set to an upper limit (0.45 N·m), a value that is close to an upper limit (4.9 kN) can be employed as the axial load. In this case, if other requirements such as a rigidity factor described later, etc. can be satisfied, there is no necessity that the value close to the upper limit must always be employed as the axial load. For instance, if radii of curvature of sectional shapes of the outer ring raceways 10a, 10b and the inner ring raceways 13, 13a are set close to 50 % of the diameter of the balls 14, 14, the rigidity factor can be secured and also the value close to the lower limit or the middle value can be employed as the axial load.

Also, the axial load is set to 1.96 to 4.9 kN and the rigidity factor is set to 0.09 or more.

Further, the running resistance (torque) of the inside seal ring 16b is regulated in a range of 0.03 to 0.2 N·m. The entering of the foreign matter such as muddy water, or the like into the spaces in which the balls 14, 14 are provided is prevented by the seal ring 16b and the cap 17 deposited onto the outer-end opening portion of the hub 7 as a sealing

member. Structures of other portions are similar to the above conventional structure shown in FIG.11. In this case, the cap 17 is fitted/fixed into the opening portion on the outer end of the hub 7 as a tight fit. A clearance is formed between the cap 17 and a top end surface (outer end surface) of the supporting shaft 4a, and thus the cap 17 never slidingly comes into contact with the supporting shaft 4a when the hub 7 is turned.

Next, FIG.2 shows a second example of the wheel supporting rolling bearing unit as the subject of the present invention. In the above structure shown in FIG.1, the inner ring 5 is fixed by the nut 6 that is screwed onto the outer end portion of the supporting shaft 4a. In contrast, in the present embodiment, the inner ring 5 is fixed to the supporting shaft 4a by pressing the outer end surface of the inner ring 5 with the caulking portion 25 that is formed by elastically deforming an outer end surface of a supporting shaft 4b as a main shaft member outward in the diameter direction. The axial load to apply the preload is adjusted by a load applied when the caulking portion 25 is worked. Structures and operations of other portions are similar to the first example.

Next, FIG.3 shows a third example of the wheel supporting rolling bearing unit as the subject of the present invention. In this example, the structure shown in above FIG.12 is varied into a structure to which the present invention can be applied.

For this purpose, in the case of the present example, an opening portion of the inner end of the outer ring 19 is closed by a cap 17a as a sealing member, and also the inner peripheral surface of the outer end portion of the outer ring 19 and the outer peripheral surface of the middle portion of the hub main body 22 are isolated from each other by the seal ring 16c. The cap 17a is fitted/fixed into the opening portion on the inner end of the outer ring 19 as a tight fit. A clearance is formed between the cap 17a and the caulking portion 25 that is formed on the inner end portion of the hub 7a as the main shaft member, and thus the cap 17a in no ways slidingly comes into contact with the hub 7a when the hub 7a is rotated. The seal ring 16d (FIG.12) between the inner peripheral surface of the inner end portion of the outer ring 19 and the outer peripheral surface of the inner ring 23 is removed. Then, the running resistance of the seal ring 16c is regulated into a range of 0.03 to 0.2 N·m. The entering of the foreign matter such as muddy water, or the like into the spaces in which the balls 14, 14 are provided is prevented by the seal ring 16c and the cap 17a. The axial load to apply the preload is adjusted by a load applied when the caulking portion 25 is worked. Structures and operations of other portions are similar to the first example. Structures of other portions are similar to the above structures in first and second examples and the above conventional structure shown in FIG.12.

Next, FIG.4 shows a fourth example of the wheel supporting rolling bearing unit as the subject of the present invention. In this example, an inner end surface of the inner ring 23 that is fitted onto the small-diameter stepped portion 24 of a hub main body 22a is pressed by a nut 30 that is screwed onto an external thread portion 39 provided to the inner end portion of the hub main body 22a as the main shaft member. A shape of a cap 17b deposited onto the opening portion of the inner end of the outer ring 19 is expanded to meet this arrangement, so that an interference between the external thread portion 39 and the nut 30 is prevented. Thus, the cap 17b by no means slidingly comes into contact with the external thread portion 39 and the nut 30 when the hub main body 22a is rotated. The axial load to apply the preload is adjusted by a torque applied when the nut 30 is tightened. Structures of other portions are similar to the second example.

Next, five examples of a particular structure of a seal ring, which can be applied to the present invention, will be explained with reference to FIGS.5 to 9 hereunder. These five examples shown in FIGS.5 to 9 show a structure that can be utilized as the inside seal ring 16b in the first and second examples of the wheel supporting rolling bearing unit shown in FIGS.1 and 2 respectively.

First, a first example shown in FIG.5 is a combinational seal ring in which an outer diameter-side seal ring 31 that

is fitted/fixed into the inner end portion of the hub 7 (FIGS.1 and 2) and an inner diameter-side seal ring 32 that is fitted/fixed onto the portion located near the inner ends of the supporting shafts 4a (FIG.1), 4b (FIG.2) are combined with each other. This seal ring has three seal lips in total, i.e., two on the inner diameter side and one on the outer diameter side. In the case of such structure, in the prior art, the torque (running resistance) required for the relative rotation between the outer diameter-side and inner diameter-side seal rings 31, 32 was 0.22 N·m or more. In contrast, in case the structure of the present example is applied to the present invention, the torque (running resistance) required for the relative rotation between the outer diameter-side and inner diameter-side seal rings 31, 32 based on the friction between top end edges of three seal lips and counter surfaces (surface of the core bar) is regulated in a range of 0.03 to 0.2 N ·m by the method described later, for example.

Next, a second example shown in FIG.6 is a combinational seal ring in which a seal ring 33 that is fitted/fixed into the inner end portion of the hub 7 (FIGS.1 and 2) and a slinger 34 that is fitted/fixed onto the portion located near the inner ends of the supporting shafts 4a (FIG.1), 4b (FIG.2) are combined with each other. The seal ring 33 has three seal lips. In the case of the present example, the torque (running resistance) required for the relative rotation between the

seal ring 33 and the slinger 34 based on the friction between top end edges of these three seal lips and the surface of the slinger 34 is regulated in a range of 0.03 to 0.2 N·m by the method described later, for example.

Next, a third example shown in FIG.7 is a combinational seal ring in which a seal ring 35a that engages with the inner peripheral surface of the inner end portion of the hub 7 (FIGS.1 and 2) and a seal ring 35b that engages with the outer peripheral surface of the portion located near the inner ends of the supporting shafts 4a (FIG.1), 4b (FIG.2) are combined with each other. In the case of the present example, this seal ring has three seal lips in total, i.e., two on the seal ring 35a engaging with the hub 7 side and one on the seal ring 35b engaging with the supporting shafts 4a, 4b side. In the case of such present example, the torque (running resistance) required for the relative rotation between the hub 7 and the supporting shafts 4a, 4b based on the friction between top end edges of these three seal lips and counter surfaces (inner peripheral surface of the hub 7, outer peripheral surface of the supporting shafts 4a, 4b, surface of the core bar) is regulated in a range of 0.03 to 0.2 N·m.

Next, in a seal ring shown in FIG.8, top end edges of two seal lips provided to a seal ring 36 that is fitted into the inner end portion of the hub 7 (FIGS.1 and 2) come into contact with the outer peripheral surface of the portion

located near the inner ends of the supporting shafts 4a (FIG.1), 4b (FIG.2). In the case of such present example, the torque (running resistance) required for the relative rotation between the seal ring 36 and the supporting shaft 4a based on the friction between top end edges of two seal lips and the outer peripheral surface of the inner end portions of the supporting shafts 4a, 4b is regulated in a range of 0.03 to 0.2 N·m.

Next, a seal ring 37 shown in FIG.9 shows a structure that can be utilized as the inside seal ring in FIG.2, and also can be utilized as a seal ring that is provided between the inner peripheral surface of the outer end portion of the outer ring 19 (FIGS.3 and 4) and the outer peripheral surface of the middle portion of the hub main bodies 22 (FIG.3), 22a (FIG.4) in the wheel supporting rolling bearing unit shown in FIGS.3 and 4 in the third and fourth examples. In this seal ring 37, three seal lips are provided to the core bar that can be fitted/fixed into the outer end portion of the outer ring 19, and the top end edges of these seal lips are brought into contact with the inside surface of the fitting flange 11a (FIGS.3 to 4) or the curved- surface portion that connects continuously this inside surface and the outer peripheral surfaces of the hub main bodies 22, 22a. In the case of such present example, the torque (running resistance) required for the relative rotation between the seal ring 37

and the surfaces of the hub main bodies 22, 22a based on the friction between top end edges of three seal lips and the surfaces of the hub main bodies 22, 22a is regulated in a range of 0.03 to 0.2 N·m.

In this event, as the method of reducing the torque required for the relative rotation between the seal ring and the counter member when any one of the above seal rings is employed, there are methods ① to ④ described as follows, for example. These methods may be employed solely or in combination respectively.

① To reduce a thickness of the seal lip.

In this case, the rigidity of the seal lip is lowered, then a contact pressure of the contact portion between the top end edge of the seal lip and the counter surface is lowered, and then the torque can be reduced.

② To eliminate substantially an interference of the top end edge of the seal lip, which is arranged closest to the internal space in which the balls are provided, out of thee seal lips.

In this case, the concerned seal lip and the counter surface constitute a labyrinth seal. Then, a frictional resistance of the concerned seal lip becomes 0.

③ To use the material having a small frictional resistance rather than the nitrile rubber that is normally used at present, as the elastic material constituting the seal

lip.

As the elastic material that has a small frictional resistance and is available in this case, as set forth in above Patent Literature 3 (JP-A-8-319379), the material obtained by mixing the plastic fine grains, which are impregnated with the lubricant, into the rubber composition constituting the seal member may be employed.

④ To devise a sectional shape of the seal lip.

As the sectional shape that is available in this case, the sectional shape in the prior invention disclosed in Patent Application No.2002-71338, for example, may be considered. In the structure in the prior invention, as shown in FIG.10, a thickness of a seal lip 38b of three seal lips 38a, 38b, 38c is reduced gradually from the base end portion toward the middle portion and then is increased from the middle portion toward the top end portion. Also, a thickness of the seal lip 38b is maximized at a part of the portion located near the top end. In addition, the shape is decided to satisfy

$$0.80d_1 \leqq d_2 \leqq 0.98d_1 \text{ and } 0.1S_2 \leqq S_1 \leqq 0.5S_2$$

where d_1 is a thickness of the base end portion of the seal lip 38b, d_2 is a thickness of a minimum thickness portion in which a thickness is minimized in the middle portion, S_1 is a sectional area of a portion extended from the base end portion to the minimum thickness portion in the axial direction, and S_2 is an area of a portion extended from the minimum thickness

portion to the top end edge.

<Examples>

Next, results of experiments to check advantages of the present invention will be explained hereunder. In first Experiment, the relationship between the running resistance (seal torque) of the seal ring as a single body and the sealing performance was obtained in five types of seal rings shown in FIGS.5 to 9. Adjustment of the sealing torque was executed by adjustment of the interference (an amount of elastic deformation) of the seal lip, adjustment of the thickness of the seal lip, exchange of the elastic material, and adjustment of the contact conditions to the counter surface. Then, six types of seal rings whose seal torque is 0 to 0.15 N·m were manufactured with respect to above five types of seal rings respectively. Then, respective seal rings were incorporated into the wheel supporting rolling bearing unit shown in FIG.1 or FIG.3 to take a muddy-water entering test. In this muddy-water entering test, as one cycle, the step of running relatively the seal rings and the member having the counter surface while pouring the muddy water on the area, in which the seal ring is provided, at a rate of 3000 cc/min was continued for 17 hours and then the step of drying the concerned area by stopping the rotation and the pouring of the muddy water was continued for 3 hours. These steps were repeated in 20 cycles every sample. Also, the lubrication of the wheel

supporting rolling bearing unit was executed by injecting a grease whose viscosity is 10 to 14 cSt (10×10^{-6} to 14×10^{-6} m^2/s). The hubs 7, 7a were revolved at 200 min^{-1} in the environment of 20 ℃.

Results of the experiment executed under such conditions are given in Table 1.

Table 1

Running Resistance [N·m]	FIG.5	FIG.6	FIG.7	FIG.8	FIG.9
0	×	×	×	×	×
0.01	△	△	×	×	△
0.03	○	○	○	○	○
0.06	○	○	○	○	○
0.10	○	○	○	○	○
0.15	○	○	○	○	○

Here, in Table 1, a mark "×" indicates the fact that a large amount of muddy water entered into the internal space in which the grease is sealed, a mark "△" indicates the fact that a small amount of muddy water entered into the same, and a mark "○" indicates the fact that the entering of the muddy water was not observed. It is appreciated based on such experimental results that, if the seal torque is in excess of 0.03 N·m, the entering of the muddy water can be prevented by all seal rings.

Next, second to fifth Experiments that were executed by incorporating the seal ring 37 shown in FIG.9 into the wheel supporting rolling bearing unit shown in FIG.4 to know the

influences of the seal torque (running resistance), the axial load to apply the preload, the rolling resistance, and the rigidity factor upon the controllability, the running torque of the overall rolling bearing unit, and the durability will be explained with reference to Tables 2 to 5. In Tables 2 to 5, a mark "×" indicates the fact that the problem arose in practical use in any respect, a mark "△" indicates the fact that the problem slightly arose in practical use in any respect, and a mark "○" indicates the fact that no problem arose in all respects. Here, second to fifth Experiments were carried out three times under the same conditions respectively.

First, Table 2 gives results of second Experiment that was executed to know the influence of the seal torque upon the running torque of the overall rolling bearing unit and the durability. This Experiment was executed at a rotational speed of 200 min^{-1}.

Table 2

Seal torque [N·m]	Evaluation		
0	×	×	×
0.01	△	×	△
0.03	○	○	○
0.06	○	○	○
0.10	○	○	○
0.15	○	○	○
0.20	○	○	○
0.33	×	×	×
0.45	×	×	×

As the result of second Experiment given in Table 2,

it was understood that, if the seal torque is in a range of 0.03 to 0.20 N·m, the satisfactory performance can be obtained in both respects of the running torque of the overall rolling bearing unit and the durability. In contrast, if the seal torque is at 0 N·m and 0.01 N·m (small), the entering of the foreign matter such as the muddy water, or the like into the internal space in which the balls 14, 14 are provided could not sufficiently prevented, and thus the problem arose in the respect of assuring the durability. In contrast, if the seal torque is at 0.33 N·m and 0.45 N·m (large), the running torque of the overall rolling bearing unit could not be suppressed low.

Next, Table 3 gives results of third Experiment that was executed to know the influence of the axial load (preload) upon the rigidity of the rolling bearing unit and the durability.

Table 3

Preload [kN]	Evaluation		
0.49	×	×	×
0.98	△	△	×
1.96	○	○	○
2.94	○	○	○
3.92	○	○	○
4.90	○	○	○
5.88	△	△	△
6.86	×	×	△

As the result of third Experiment given in Table 3, it was understood that, if the axial load is in a range of 1.96

to 4.90 N·m, the satisfactory performance can be obtained in both respects of the controllability and the durability of the rolling bearing unit. In contrast, if the axial load is at 0.49 kN and 0.98 kN (too small), the rigidity of the rolling bearing unit was low and thus the sufficient controllability could not be assured. In contrast, if the axial load is at 5.88 kN and 6.86 kN (too large), the rolling resistance was increased and the durability of the rolling bearing unit was degraded.

Next, Table 4 gives results of fourth Experiment that was executed to know the influence of the rolling resistance upon the rigidity of the rolling bearing unit and the durability. This Experiment was executed at a rotational speed of 200 min^{-1}.

Table 4

Rolling Resistance [N·m]	Evaluation		
0.10	×	×	×
0.12	×	△	△
0.15	○	○	○
0.25	○	○	○
0.35	○	○	○
0.45	○	○	○
0.55	△	×	△
0.65	×	×	×

As the result of fourth Experiment given in Table 4, it was understood that, if the rolling resistance is in a range of 0.15 to 0.45 N·m, the satisfactory performance can be obtained in both respects of the controllability and the durability of the rolling bearing unit. In contrast, if the

rolling resistance is at 0.10 N·m and 0.12 N·m (too small), the rigidity of the rolling bearing unit was low and thus the sufficient controllability could not be assured. In contrast, if the rolling resistance is at 0.55 N·m and 0.65 N·m (too large), the durability of the rolling bearing unit was degraded.

In addition, Table 5 gives results of fifth Experiment that was executed to know the influence of the rigidity factor upon the rigidity of the rolling bearing unit.

Table 5

Rigidity factor	Evaluation		
0.07	×	×	×
0.08	×	△	×
0.09	○	○	○
0.10	○	○	○
0.15	○	○	○
0.18	○	○	○

As the result of fifth Experiment given in Table 5, it was understood that, if the rigidity factor is 0.09 or more, the satisfactory performance of the controllability can be obtained. In contrast, if the rigidity factor is at 0.07 and 0.08, the rigidity of the rolling bearing unit was low and thus the sufficient controllability could not be assured. In this case, as described above, the higher rigidity factor is better insofar as other requirements are satisfied.

Next, Table 6 gives results of Experiment that was executed to know the influences of the seal torque and the

rolling resistance upon the running torque of the rolling bearing unit as a whole. This Experiment was executed at a rotational speed of 200 min^{-1}.

Table 6

Rolling Resistance [N·m] \ Seal torque [N·m]	0.15	0.2	0.33
0.35	○	○	△
0.45	○	○	×
0.55	△	×	×

In Table 6, a mark "×" indicates the fact that the running torque was large as a whole, a mark "△" indicates the fact that the running torque was slightly large, and a mark "○" indicates the fact that the running torque was small. As apparent from this Table 6, the present invention in which both the seal torque and the rolling resistance are suppressed to 0.2 N·m or less and 0.45 N·m or less respectively could suppress as a whole the running torque low such as 0.65 N ·m or less.

Four examples of the specifications of the wheel supporting rolling bearing unit belonging to the technical scope of the present invention are given in Table 7.

Table 7

No	Preload [kN]	Rolling torque [N·m] (measured at 200 min^{-1})	Rigidity factor	Ball Diameter[mm]	PCD[mm]	Inter-row distance[mm]	Contact angle[deg]
1	1.96	0152	0.093	ϕ9.525	46	24	40
	4.9	0.448	0.102	ϕ9.525	46	24	40
2	1.96	0.150	0.091	ϕ12.7	51	35	40
	4.9	0.445	0.100	ϕ12.7	51	35	40

In Table 7, the ball diameter denotes a diameter of the ball, the PCD denotes a pitch diameter of a ball sequence of these balls, the inter-row distance denotes a pitch (center distance between the balls) of the ball sequences arranges in double rows in the axial direction, and the contact angle denotes a contact angle between the balls and the inner ring raceway and the outer ring raceway.

Also, the influence of the contact angle upon the rigidity factor is given in Table 8. It was appreciated from Table 8 that the rigidity factor becomes small as the contact angle becomes small.

Table 8

Preload [kN]	Contact angle [deg]	Rigidity factor	Ball Diameter [mm]	PCD [mm]	Inter-row Distance [mm]
1.96	40	0.093	ϕ 9.525	45	24
1.96	35	0.089	ϕ 9.525	45	24

The present invention is explained in detail with reference to particular embodiments, but it is apparent for the person skilled in the art that various variations and modifications can be applied without departing from a spirit and a scope of the present invention.

The present application was filed based on Japanese Patent Application (Patent Application No. 2002-261195) filed on September 6, 2002 and the contents thereof are incorporated herein by the reference.

<Industrial Applicability>

Since the wheel supporting rolling bearing unit of the present invention is constructed and acts as described above, such bearing unit can contribute to the improvement in the running performances of the vehicle, mainly the controllability, the acceleration performance, and the fuel consumption performance, by reducing the running torque of the hub, which rotates together with the wheel, while assuring the rigidity and the durability.

An example of a trial calculation to improve the fuel consumption performance will be explained hereunder. The running resistance of the wheel supporting rolling bearing unit having the above structure shown in FIGS.1 to 4 was almost 1 N·m in the prior art. In contrast, the running resistance of the wheel supporting rolling bearing unit of the present invention is in a range of 0.18 to 0.65 N·m. That is, the running resistance is lowered by 35 % or more rather than the prior art. It is considered that, if the running resistance of the wheel supporting rolling bearing unit is lowered by 10 %, the fuel consumption (fuel consumption ratio) can be improved by about 0.1 %. Therefore, suppose that the car the fuel consumption of which is about 10 km/L travels 100,000 km a year, the fuel can be saved by about 35 to 82 L a year by employing the wheel supporting rolling bearing unit of the present invention. Suppose that 1,000,000 cars travel in this

country, the fuel that can be saved a year reaches 35,000,000 L to 82,000,000 L. In addition, it is possible to say that the industrial usability is extremely high because the fuel consumption can be improved not to cause other disadvantages.